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(72) Inventor: RAYMOND SHARPE

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(54) VARIABLE SPEED TRANSMISSION SYSTEMS

(71) We, LUCAS INDUSTRIES LIMITED, a British Company of Great King Street, Birmingham B19 2XF, England, do hereby declare the invention for which we pray that a Patent may be granted to us and the method by which it is to be performed, to be particularly described in and by the following statement:-

This invention relates to variable ratio frictional drive gears of the kind comprising basically two axially spaced torus discs or rotors one serving as an input and the other an output between which there is a set of circumferentially spaced drive rollers in frictional rolling contact with part toroidal surfaces on the discs, each roller being rotatably mounted in a bearing structure which can tilt about an axis at right angles to the axis of rotation of each roller so as to vary the distances from the gear axis at which the roller engages the two discs respectively, thus varying the drive ratio of the gear. The angle of tilt of the roller bearing structure as it controls the drive ratio of the gear, is called the ratio angle.

One way of changing the ratio angle is to provide means to apply a force to each of the roller bearing structures to move it generally tangentially with respect to the gear axis, and by allowing the rollers then to steer themselves towards a different ratio angle. The rollers are each mounted in their bearing structures in such a way that they are inclined at an angle to a plane perpendicular to the gear axis. This angle is called the caster angle. Gears of this general construction are referred to as gears with tangentially controlled roller bearing structures. There is furthermore provided fluid pressure operated means for loading the discs axially so that there is pressure between the toroidal surfaces and the surfaces of the rollers engaging with them. Such a drive gear will for convenience herein be described as being of the kind specified.

End loading, as such axial pressure is usually called must be substantial and is generally applied hydraulically. Furthermore the higher the speed and the higher the load, being transmitted through the transmission system the higher the pressure must be in order to maintain frictional contact between the rotor toroidal and roller surfaces, in order, in turn, to maintain efficiency of load transmission. The frictional contact may be through the agency of lubricating fluid which in this type of apparatus is usually referred to as tractant fluid.

The input must rotate in the direction in which it tends to drag each roller against the control force which controls the tangential position of the rollers. The caster angle must be such that each roller tilt axis is inclined away from the input disc in the direction of movement of the disc. This criterion arises out of the fact that stable operation at any given ratio angle occurs when the axis of rotation of each roller passes through the gear axis. Unless the caster angle is in the sense just referred to, tangential displacement of a roller (by virture of an increase or decrease in the load on the gear or in controlling fluid pressure will result in the torus discs producing a steering force on the roller which will tilt the roller in the direction opposite to that which is required to move the roller axis back to intersect the gear axis, so that the roller will be moved away from, instead of towards, its new stable position.

This invention is particularly though not exclusively concerned with gears in which the plane of each roller, normal to the axis of rotation of the roller and passing through the points of contact of the roller with the two opposed torus discs, contains the axis about which the roller tilts, being tangential to the torus centre circle (i.e. the locus of the centre of the circle revolved to generate the torus) as distinct from gears in which the

same plane for each roller is closer to the main axis of rotation of the gear.

It is the object of the invention to provide a variable speed transmission of the kind specified having fluid pressure operated means for applying axial loading to apply pressure between the toroidal surfaces of the rotors and the surfaces of the rollers engaging them, to provide efficient transmission of loads through the system.

According to the present invention there is provided a variable speed transmission system of the kind specified in which said fluid pressure operated means for loading the rotors axially comprises means defining two cavities in which fluid is admitted to exert said axial pressure, said cavities being arranged to provide respective axial loading in tandem.

The invention will now be described by way of example with reference to the accompanying drawings in which the single Figure is a cross-sectional view showing a transmission system constructed in accordance with the invention.

The transmission system is principally designed for use in driving aircraft accessories and in particular an alternator. The alternator is driven from an aircraft main engine but is required to be rotated at constant speed. The transmission is therefore described for variable input speed, but constant output speed. It is however, to be understood that transmission incorporating the invention as herein defined can be used in transmissions of this sort with other operating characteristics including constant input and variable output speed and variable input as well as output speeds.

Referring to the drawing, the general layout of the transmission is illustrated. The system inludes a variable ratio drive unit having three rotors 10, 11, 12 which have respective part toroidal surfaces 10*a*, 11*a* and 12*a* and 12*b* respectively. The rotor 12 is situated mid-way between the rotors 10 and 11, and is provided with its part toroidal surfaces 12*a*, 12b, on opposite axially presented sides thereof. The rotor 10 has its part toroidal surface 10*a* presented towards the surface 12*a*, and similarly the surface 11*a* of the rotor 11 is presented towards the surface 12*b* of the central rotor 12. The rotors 10, 11 are input rotors and the rotor 12 is an output rotor. However, the system will operate perfectly satisfactorily with the rotors 10, 11 as output and the input is the rotor 12. Situated between the rotors 10, 12 and 11, 12 are respective sets of flat rollers 13, 14. These are rotatable in a manner which will be described and are for this purpose carried in respective bearings 15, 16. The rollers are shown in Figure 1 in positions in which they engage the respective surfaces 10*a*, 12*a* and 11*a* 12*b*, at different distances from the axis of rotation of the rotors 10, 11, 12. Such axis is identified at 17. The rotors 10, 11 are carried non-rotatably upon a hollow shaft 18. This is supported on suitable fixed structure 22 by means of bearings 19, 20 situated near its opposite ends respectively.

The input rotor 10 has on its external periphery, gear teeth 23, engaging with a gear ring 24, on a hollow stepped shaft 25. This hollow stepped shaft is mounted for rotation about an axis 26, parallel with the axis 17. Connecting the hollow stepped shaft 25, with a surrounding sleeve 27, is a one way clutch 28 which is described and claimed in the Complete Specification of Patent Application No. 33908/76. (Serial No 1600975). The sleeve 27 has gear teeth 29, meshing with a gear (not shown) which drives auxiliary equipment which forms no part of this invention.

The output rotor 12 has external gear teeth 30 and this represents the output of the drive unit.

Driving the shaft 18, though gear teeth 34, thereon is a gear wheel 35 which is carried on a further hollow sleeve 36. Between the sleeve 36, and an input shaft 37, with at one end, dogs 38 is a coupling incorporating an intermediate slidable sleeve 39 and an element 49 which is arranged to melt and allow the sleeve 39 and hence the shaft 37 to move under the influence of springs 31 in the event of this part of the system reaching a temperature in excess of a predetermined value to disconnect the input drive from the system. This forms the subject of co-pending British Patent Application number 33909/76. (Serial No 1600976).

Adjacent to the bearing 20 on the shaft 18, which is supported in the fixed structure 22, is an end load device for applying load to the input rotor 11, this load being transmitted through the set of rollers 14, to the rotor 12, and thence to the set of rollers 13 to the rotor 10. A backup ring 10*b* with a spart spherical surface prevents any axial movement of the rotor 10 which is provided with a corresponding part spherical surface to mate with the backup ring 10*b*. To produce the end loading on the rotor 11, the latter carries a circular casing 41 secured in place by screws 42. The casing and rotor 11 define between them two cavities 43, 44. Within these are respective annular pistons 45, 46 carrying external seals 49, 50. The seal 49 engages, on a cylindrical surface within the cavity at the side of the rotor 11 remote from its part toroidal surface 11*a*. The seal 50 engages on a co-axial cyldindrical surface of a component 51 fixed within the casing 41 and having an annular portion which defines the division between the two cavities 43 and 44. At its minor annular periphery it has a

further cylindrical surface on which engages a further seal 52 which acts on an external cylindrical surface on the piston 45.

Between the bearing set 20, and the piston 46, is a thrust ring 53. An end of the piston 46 abuts against the thrust ring 53. This ring carries a disc spring 54 of bowed configuration, the face of which adjacent its external periphery engages on the outside surface of the casing 41. Further seals 55, 56 are provided between the interior of the rotor 11, and the shaft 18, and between the piston 46, and the shaft 18.

The shaft 18, has a central coaxial drilling 48, leading from the end at which the bearing 20 is provided with a rotary fluid joint 21. There is series of further axial drillings 47, one in communication with the hollow opposite end of the shaft 18. The drillings 47, of which there are two in order to balance the shaft overlap the drilling 48 without communicating with it. One of the drillings 47 communicates at its inner end with a cross drilling 57, which in turn opens into slots 58 in the axial end face of the thrust ring 53 with which the piston 46 abuts. These in turn open into the cavity 44.

At its outer periphery the component 51 is spaced from the interior of the casing 41 over part of its length to define an annular space 68. Communicating with this space and extending through the component 51 are drillings 64 which thus provide communication between the outer zones of the two cavities 43, 44. The hollow interior of the shaft 18 is supplied with fluid at low pressure and the drillings 47, cross drilling 57, slots 58, are open to the same source of low pressure fluid.

The portions of the cavities 43, 44 at one side of each of the pistons 45, 46 are therefore also subjected to this pressure through the communicating space 68 and drillings 64. Through small outlets 59 in the wall of the shaft 18, this fluid is also fed to the space surrounding the rotors and rollers to act as lubricant in this region. Such fluid is a tractant fluid which is capable of withstanding the very high pressures exerted between the mating surfaces of the rotors and the rollers and which furthermore ensures efficient transmission of torque through the surfaces.

This fluid at relatively low pressure is supplied through a connection in the open end of the shaft 18. This connection comprises a plug 60 in which is rotatably supported a supply pipe 61.

The inner end of the drilling 48 has a cross drilling 62, which in turn communicates with slots 63, in the end of the piston 46, abutting the piston 45 to enter the cavity 44 at the side of the piston 46 adjacent to the component 51 this zone being indicated by numeral 66.

Slots 65, between the shaft 18 and the piston 45 provide communication between the cross drilling 62 in the shaft 18, and the cavity 43 at the side of the piston 45 adjacent to rotor 11. The fluid supply through the joint 21 and the drilling 48, the cross drilling 62 and the slots 63 and 65 is at high pressure. The zone in the cavity 44 to which the fluid is admitted is identified by numeral 67.

This high pressure thus exerts against the pistons 45, 46 and the rotor 11 and components 51, a force sufficient to load the rotors and rollers. However, since the rotor 11 casing 41 and component 51 are rotating, in use, at high speed the fluid in the two zones 66, 67 is subject to centrifugal force so that a pressure gradient exists radially of these zones, with the highest pressure existing at the outer peripheries of the zones 66, 67.

At the other side of each of the pistons 44, 45 the lower pressure fluid is however also subject to centrifugal force of equal magnitude and of the same gradient radially of the cavities. The effects of these centrifugally generated pressure gradients are however of substantially equal opposing effect at opposite sides of each of the pistons 44, 45 so that the effective pressure or force applied axially of the system to the rotor 11 is substantially that which the fluid pressure would apply if the centrifugal effect were not present, that is the pressure existing if the rotor 11, casing 41 and component 51 were not rotating.

The rollers 13, 14 are shown in the drawing at respective opposite angles to a plane perpendicular to the axis 17 of the shaft, and their points of contact are at different radial distances on the surfaces 10*a* and 12*a* and similarly on the surface 11*a* and 12*b* from the axis 17.

Such angular displacement which is adjustable, produces change in the speed of the output rotor 12 relatively to that of the rotors 10, 11. The angular setting of the rollers to regulate the relative speed ratio within the required conditions is achieved by hydraulically changing the positions of the rollers 13, 14 in a direction generally tengential with respect to lines of contact between the rollers and the rotor part toroidal surfaces. The hydraulic actuation is typically of the kind described in pending Patent Application No. 33904/76. (Serial No 1600972).

The hydraulic pressure available for the adjustment of the rollers is obtained from the same source as that supplied through the drilling 48 in the higher pressure zones of the end load device.

WHAT WE CLAIM IS:-

1. A variable speed transmission system of the kind specified in which said fluid pressure operated means for loading the rotors axially comprises means defining two

cavities in which fluid is admitted to exert said axial pressure, said cavities being arranged to provide respective axial loading in tandem.

2. A transmission system as claimed in claim 1 in which there is common fluid pressure feed to the two cavities and fluid is also admitted to further cavities at lower pressure than that obtaining in the said two cavities

3. A transmission system as claimed in claim 1 or claim 2 in which the means defining said cavities includes rotatable parts whereby the fluid is subjected to centrifugal forces.

4. A variable speed transmission system substantially as hereinbefore described with reference to and as shown in the accompanying drawing.

MARKS & CLERK,
Alpha Tower,
ATV Centre,
Birmingham, B1 1TT.
Agents for the Applicants.

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1600974 COMPLETE SPECIFICATION

1 SHEET

This drawing is a reproduction of the Original on a reduced scale